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(54) Synchronizer for positive clutches

(57) A synchronisation arrangement comprises a synchronising body member (1) with external clutch teeth which is rigidly connected to a shaft, a displaceable shift sleeve (3) axially movable with respect to the body (1) and having internal clutch teeth (4) engageable with external clutch teeth on a gear wheel (Z1, Z2) arranged concentrically therewith and at least one synchronising ring (2) disposed between the gear wheels (Z1, Z2) and having a friction surface which cooperates with a friction surface on a gear wheel (Z1, Z2). Shift elements in the form of locking and pressure levers are disposed between the body (1) and the sleeve (3) to transmit the axially orientated shift force from the shift sleeve to the synchronising ring (2) and simultaneously lock the shift path of the shift sleeve until the body (1) and the gear wheel are synchronised. Each pressure lever is mounted in the body (1) so that it may not be displaced in a radial manner and may perform a free limited pivoting movement in all directions.

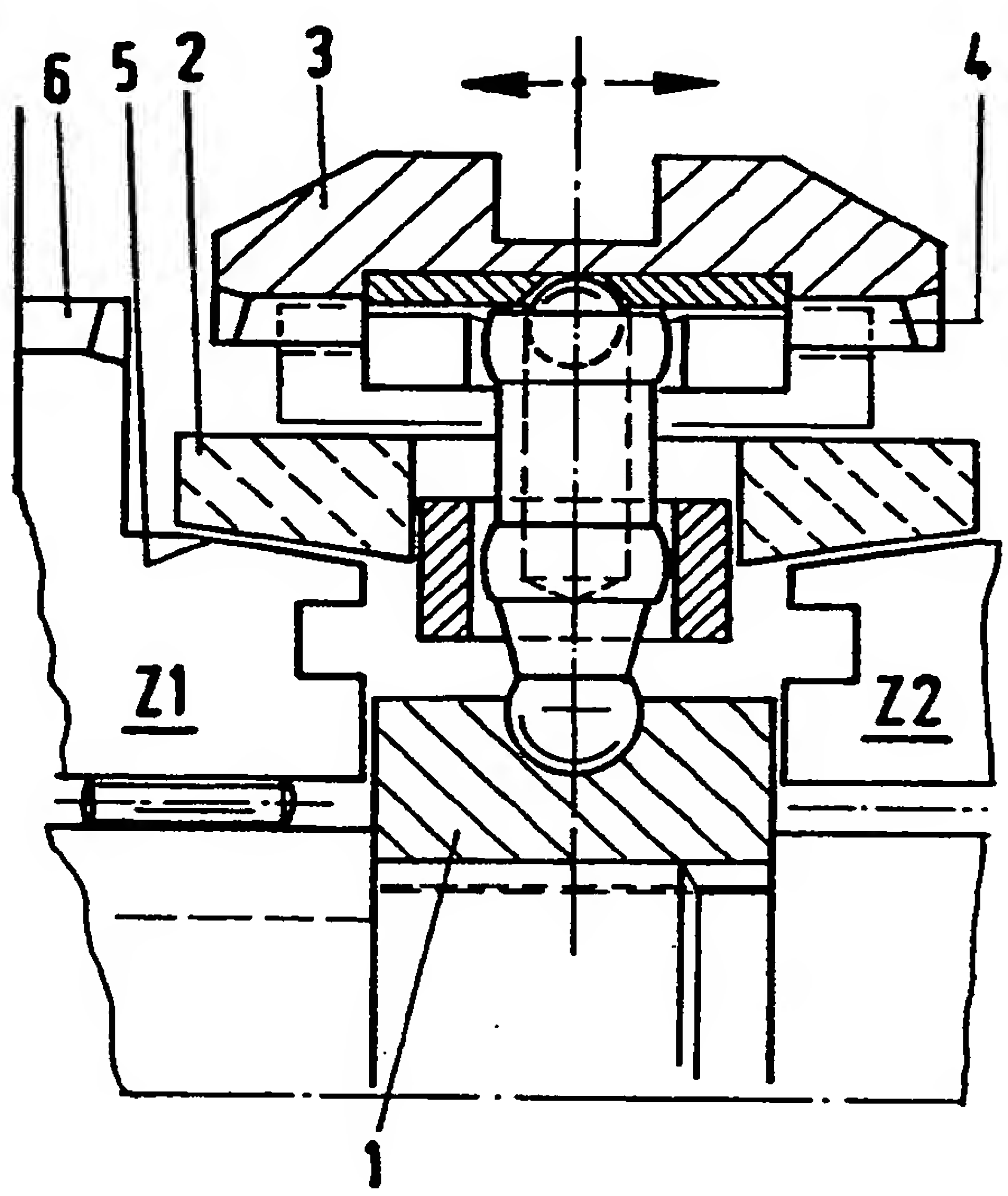

FIG. 2

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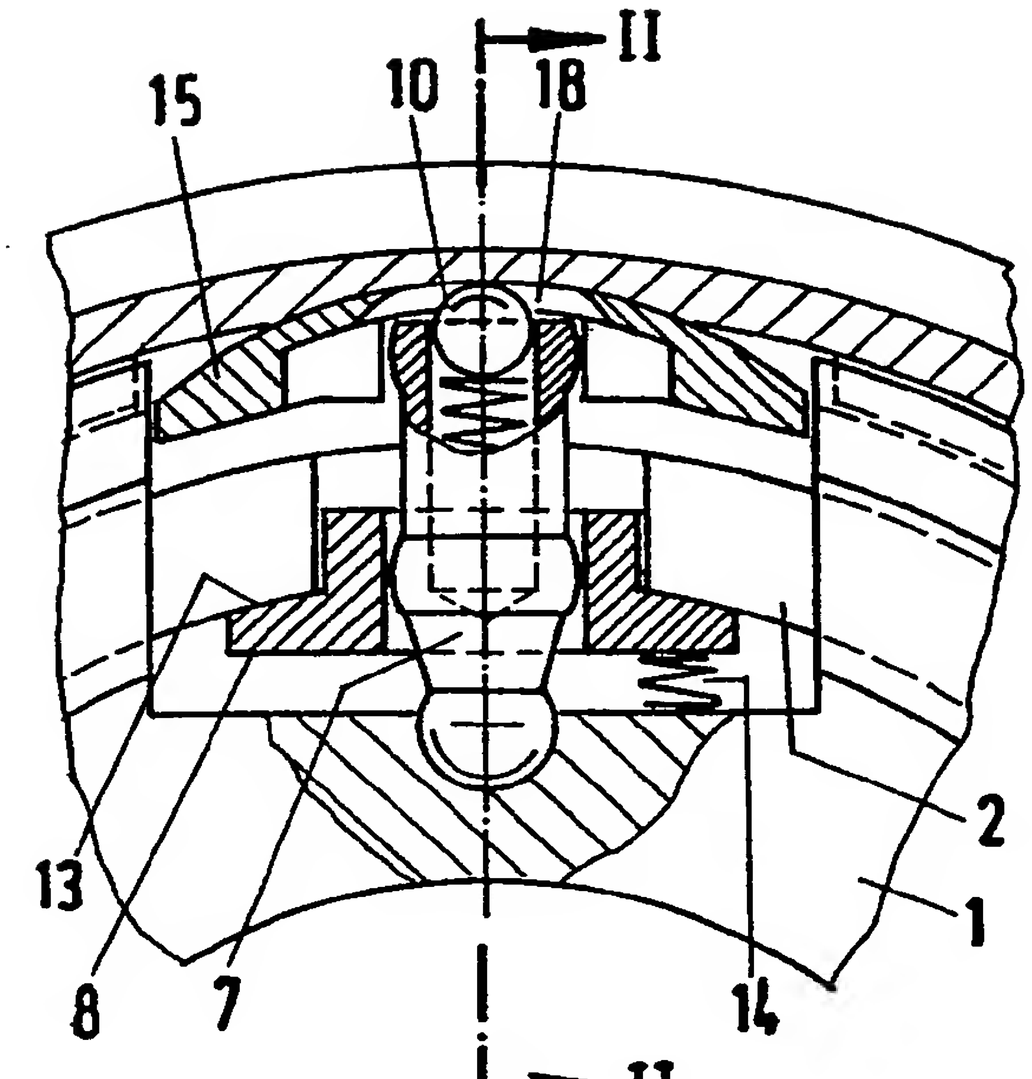

FIG.1

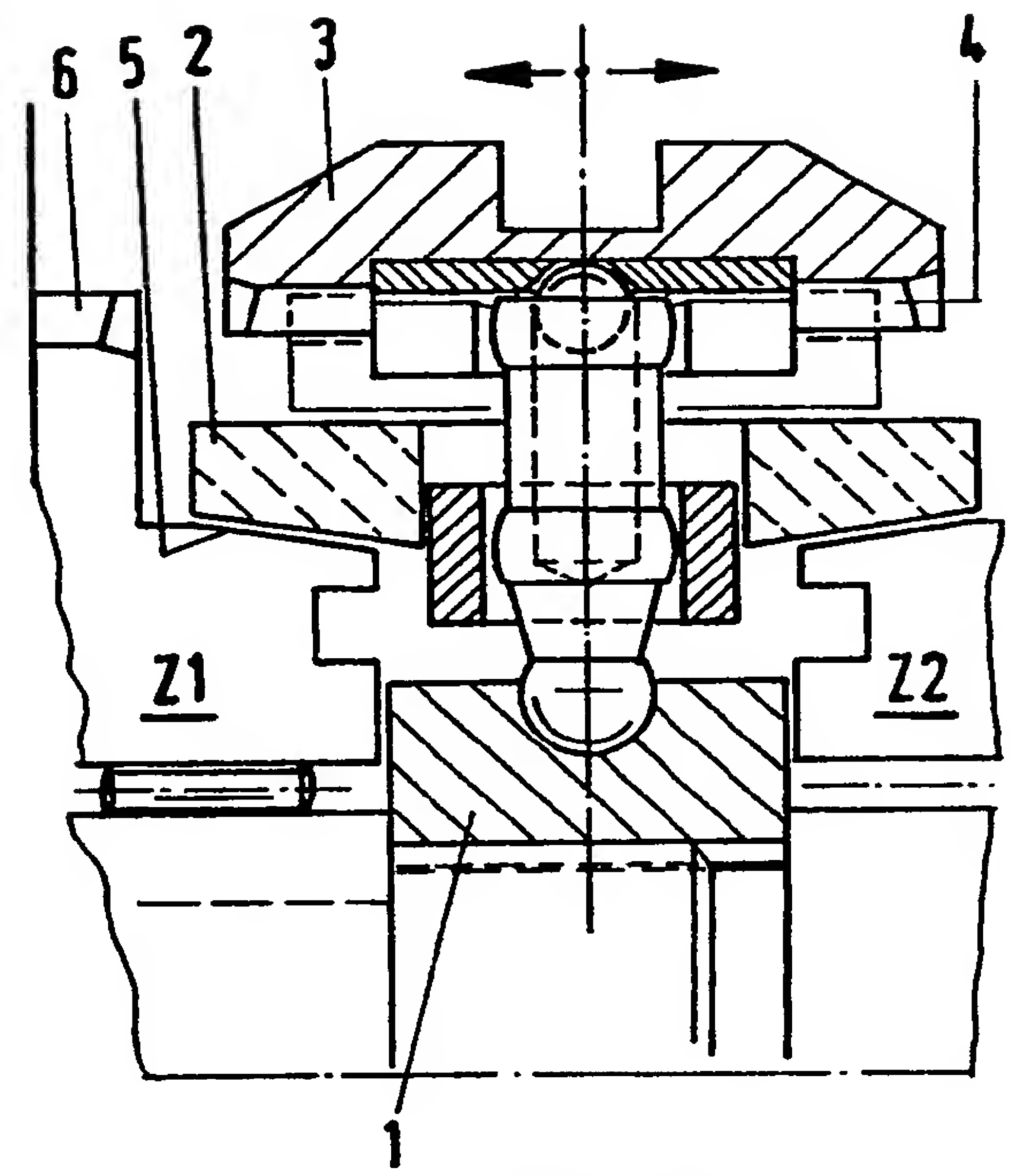

FIG.2

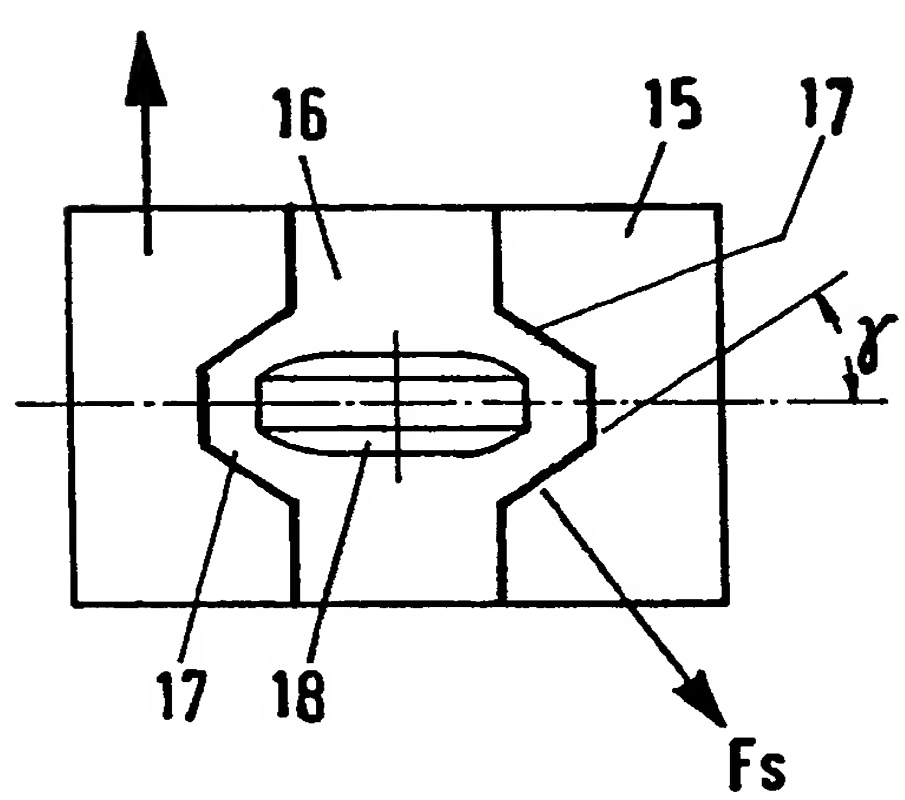

FIG.3

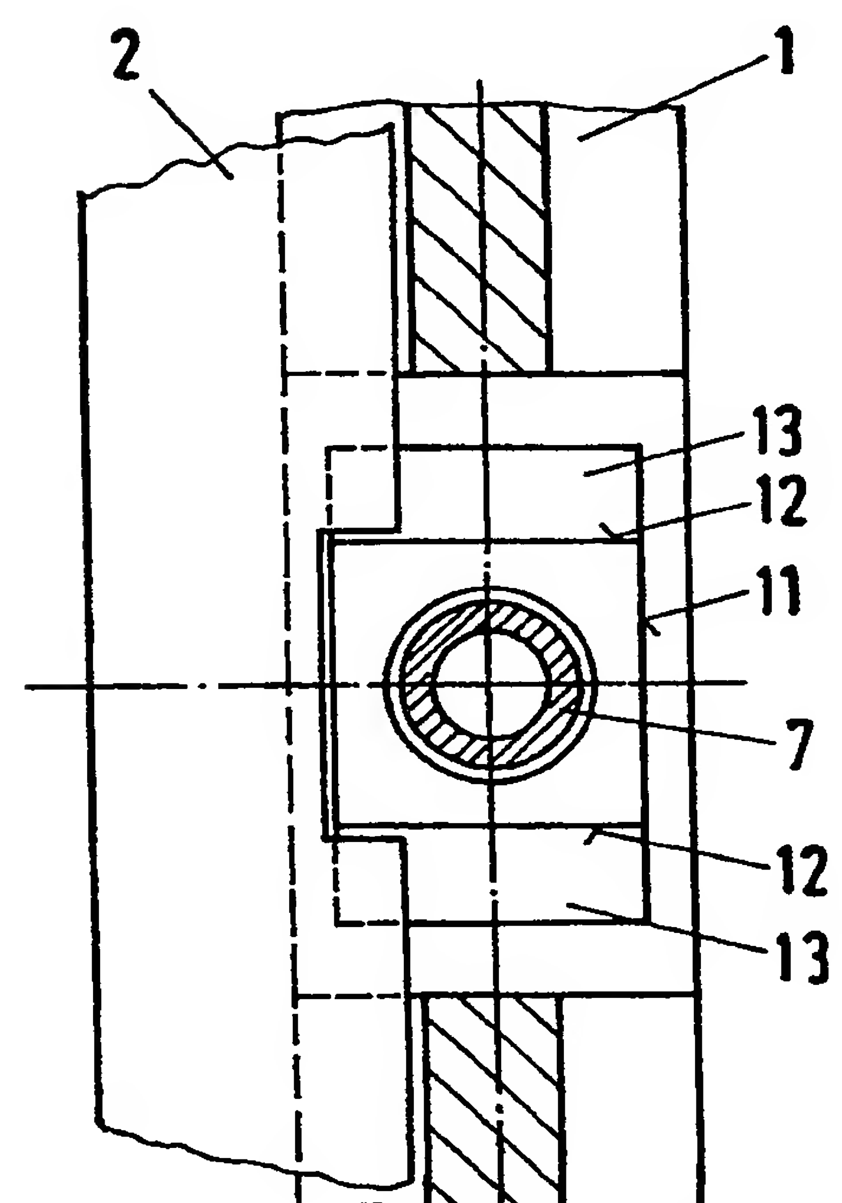

FIG.4

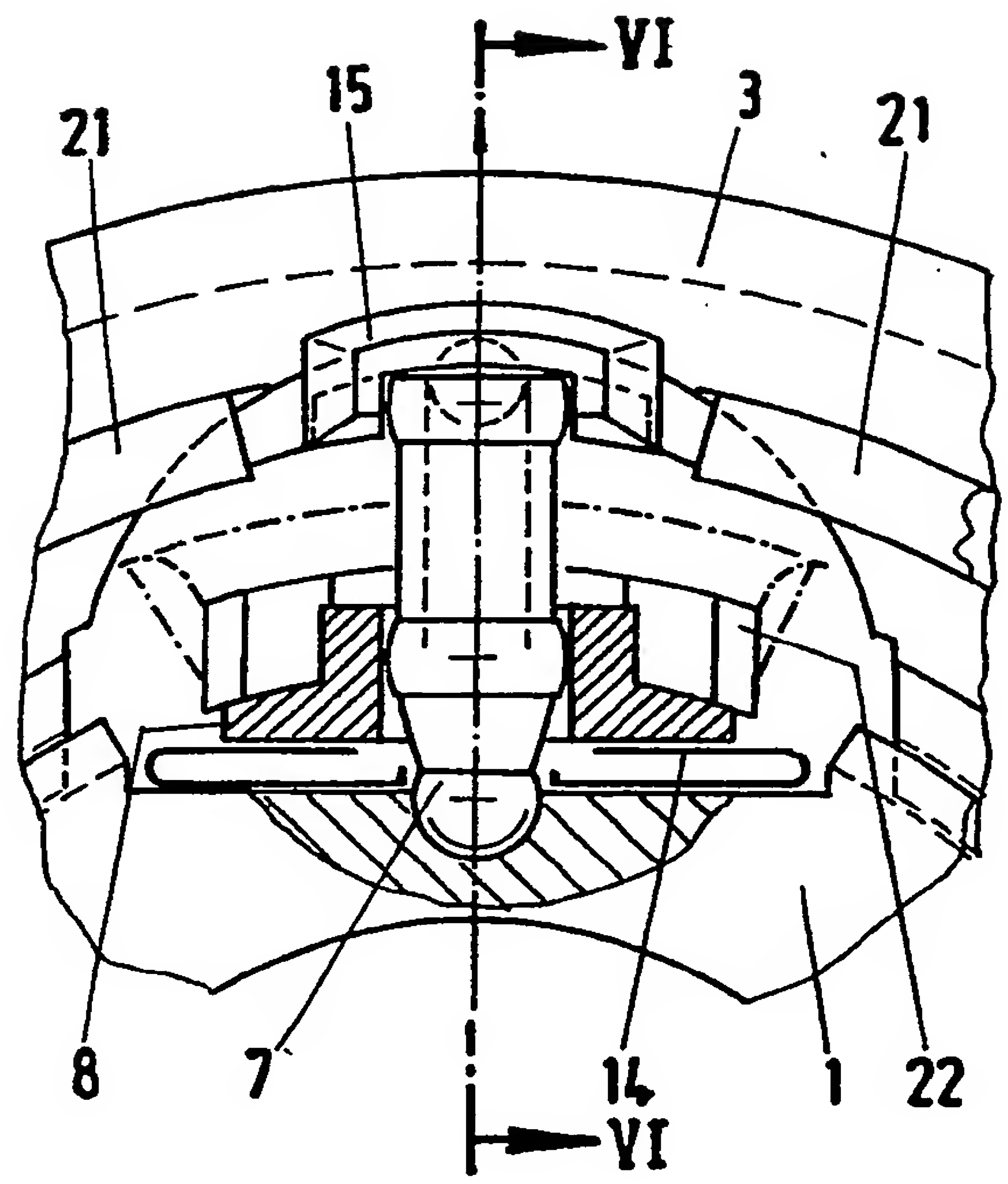

FIG. 5

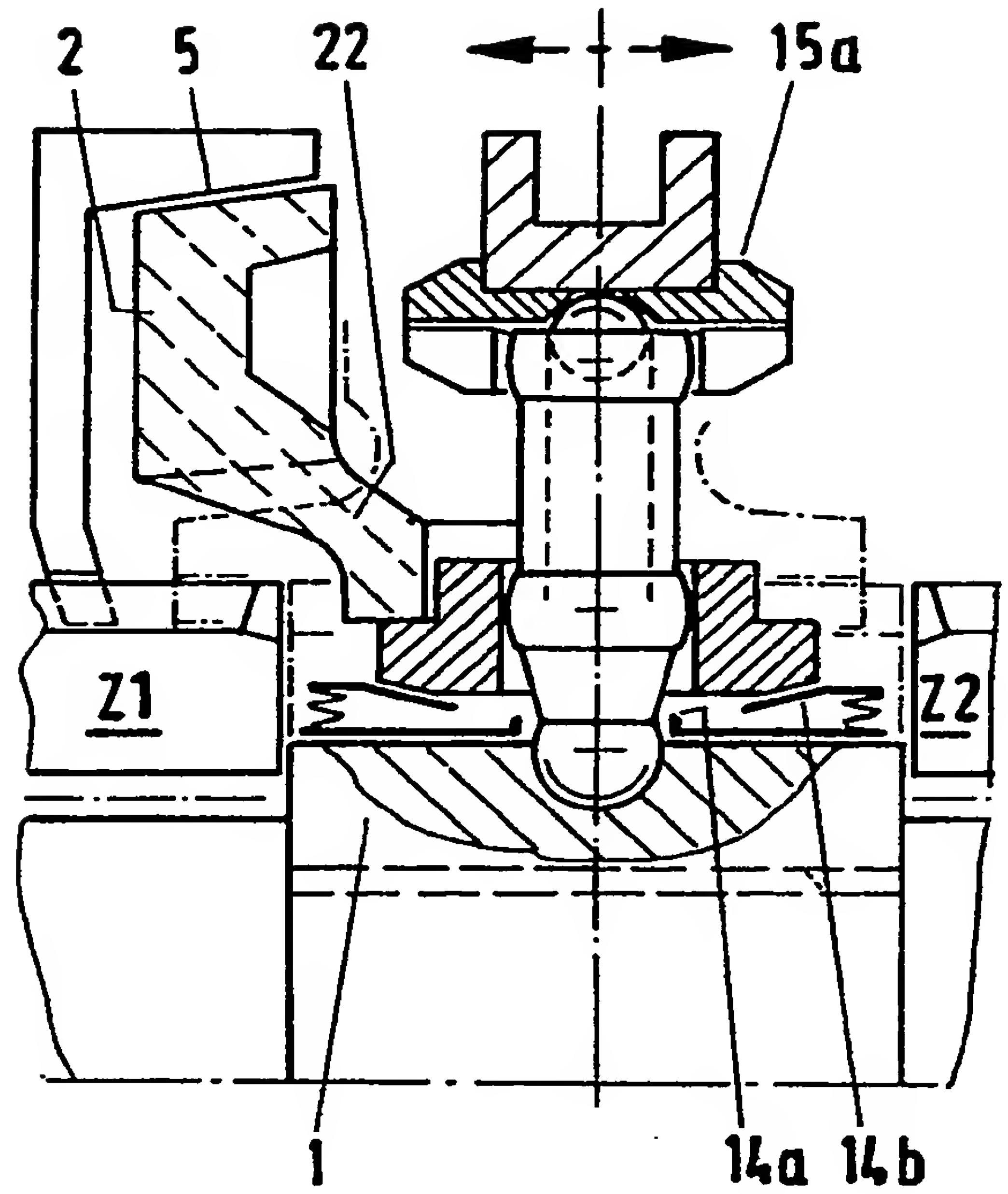

FIG. 6

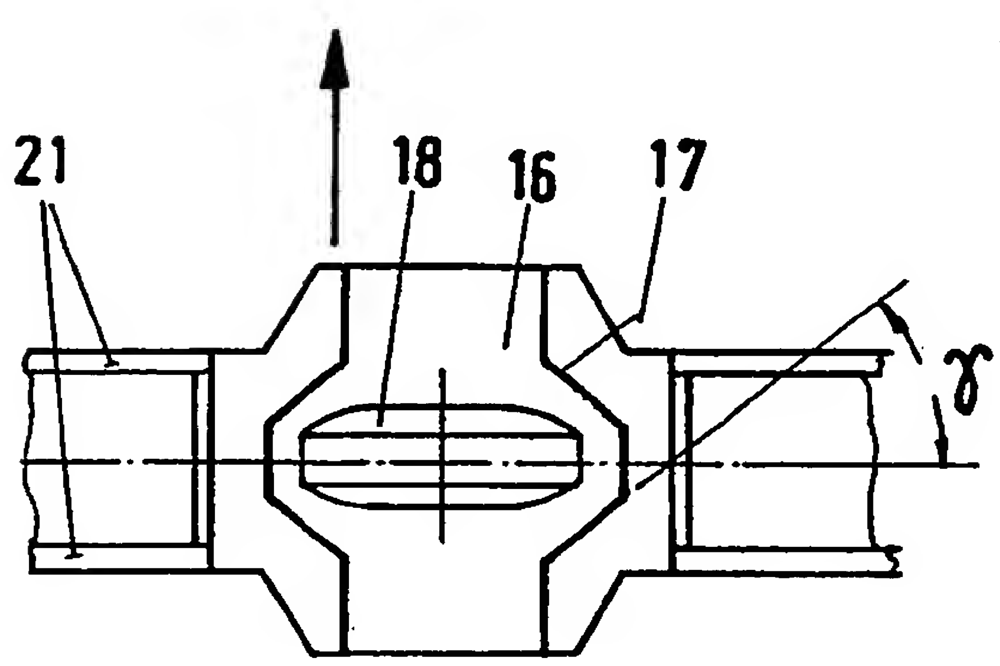

FIG. 7

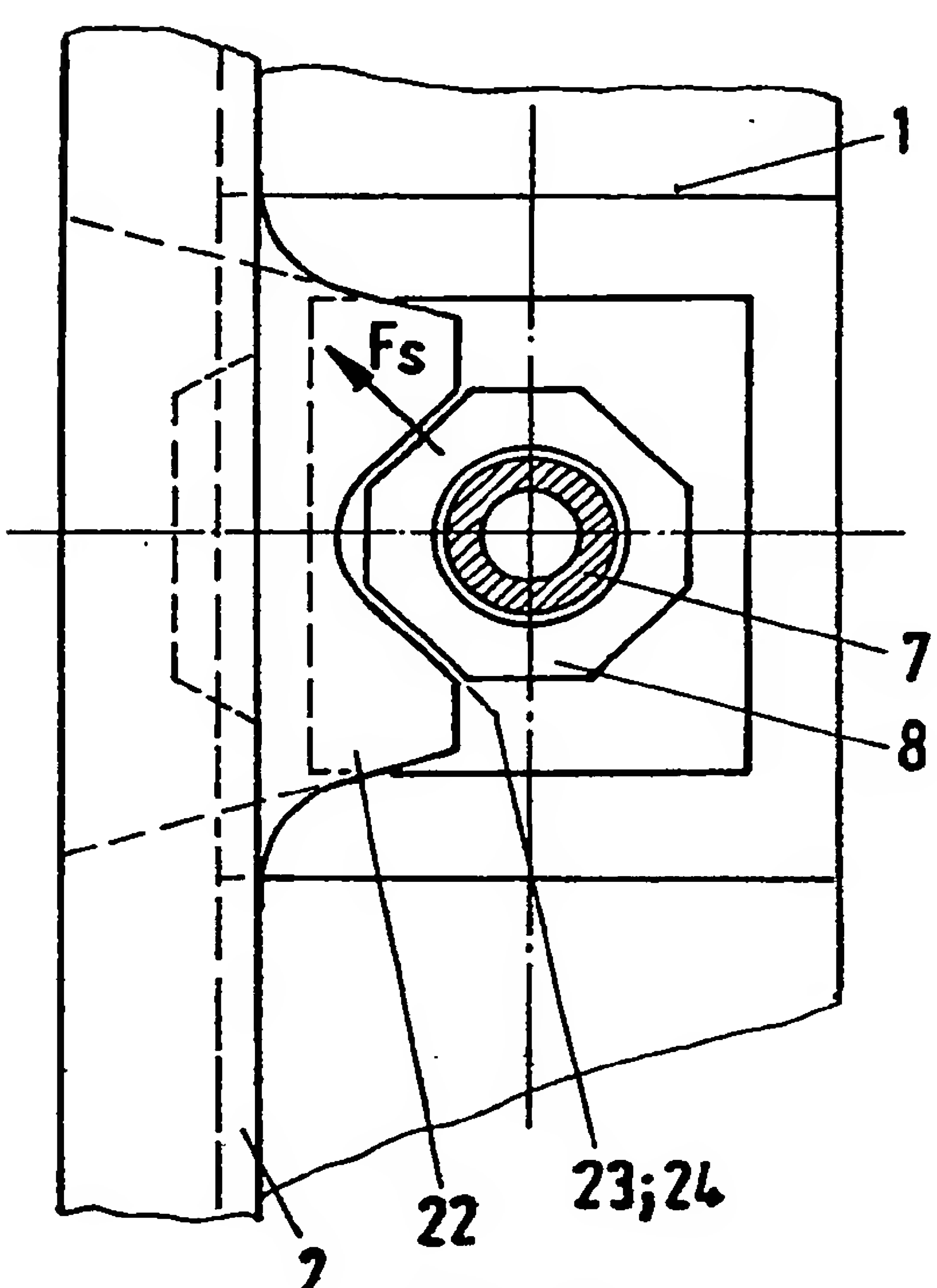

FIG. 8

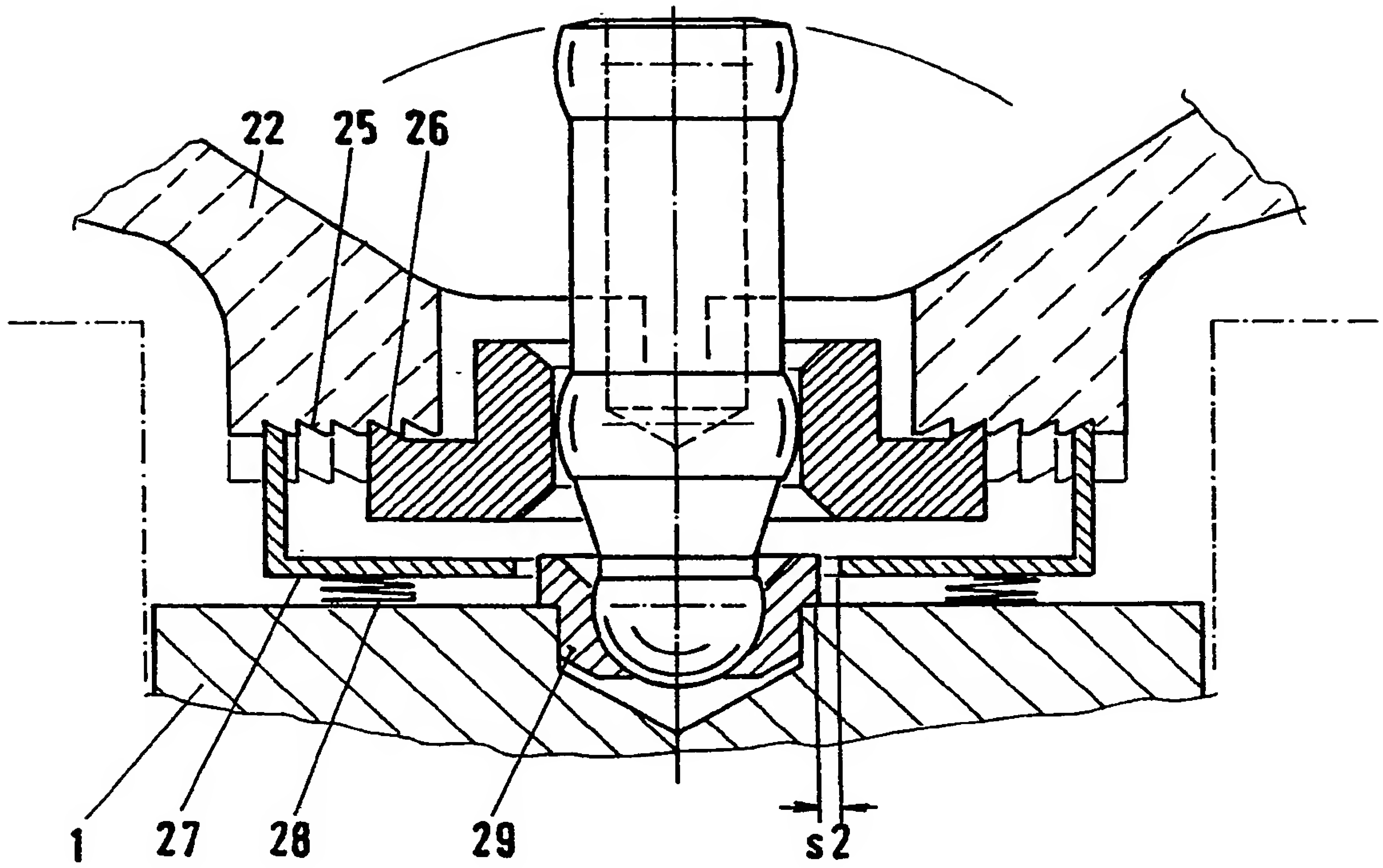

FIG.9

SPECIFICATION

Locking synchronisation arrangement for gear shifts

This invention relates to a locking synchronization arrangement for gear shifts wherein two concentrically disposed components such as a gear shaft and a gear wheel rotating at different speeds are firstly synchronized by means of a friction clutch and are then positively connected in the direction of rotation.

Locking synchronization arrangements are almost exclusively used in gear shifts for land vehicles in which the speed and torque of the prime mover must frequently be adapted to different load conditions.

There are several constructions of this type which are successful in practice, all of which, however, operate in accordance with the same principle. When the gear ratio is changed, two gear components rotating at different speeds – in most cases a shaft and a gear wheel – must in the first instance be brought to the same speed (i.e. synchronized) and then positively coupled together. The axially movable portion of the friction clutch operates, during the process of synchronization with the friction torque, directly as a servomotor of a locking device which prevents a positive connection as long as there is a speed difference.

The physical and mathematical relationships form part of the general state of the art so they will not be discussed in detail here.

If the design and material values of the synchronising clutch as given in the factor k are combined, the following simplified relationship applies to the axial coupling-contact force (i.e. shifting force) required during the synchronization:

$$F_a = \frac{\mathcal{J} \times \Delta\omega \times k}{t} \quad (N)$$

$\mathcal{J}$ (kgm^2) = Inertia moment of the rotating parts whose speed is modified,

$\Delta\omega$ (sec^{-1}) = Difference in angular velocity,

t (sec) = Time taken to reach synchronism.

If the product $F_a \times t$ (shifting force x shifting time) is designated as a shifting movement, it is then possible to state that the shifting movement increases proportionally to the rotating mass and to the size of the gear change. On average, there is an increase of the shifting movement with increasing drive power and increasing drive torque. In the case of operating gears (cars, lorries and tractors) which are changed manually to a predominant extent in the current art, it is necessary, with respect to simple and more reliable operation, for the gear change to take place as rapidly as possible with the minimum of energy consumption, i.e. that the shifting movement should have the smallest possible value.

In the case of current, mass-produced gear shifts, the manual shifting force supplied to the gear shift sleeve via the shift fork operates directly, during the process of synchronization, as an axial contact force on the synchronizing coupling.

However, in our British Patent Specification No. 2,048,399 there is disclosed an arrangement in which radial pressure elements are disposed between the shift sleeve and the synchronizing coupling, which elements have the effect of a step-up lever and simultaneously act as shift path locking elements. It is therefore possible, with a corresponding construction, to produce an axial contact force acting on the synchronizing ring which is greater than the shifting force applied from outside on the shift sleeve. The effect is to a certain degree an "internal shifting force ratio" which is only effective during the synchronization process and with which it is possible to achieve a desired reduction of the shifting movement. This design however has certain drawbacks:–

a) The power flow from the shift sleeve to the synchronizing ring moves over a number of contact surfaces on the pressure element and in addition produces internal force components (as a result of several inclined surfaces). In order to overcome the locking effect, the pressure element must be pushed radially inwardly. A linear sliding movement takes place at all the contact points with a frictional resistance. This frictional resistance increases with an increasing lever transmission ratio of the pressure element and leads to an automatic locking effect. In order to commence the synchronization process, an initial contact load must firstly be applied to the synchronizing ring, which load is produced by the radial initial spring tension of the pressure element. This initial load must be overcome during change over (i.e. production of the positive connection). The centrifugal force produced by the pressure element also overlaps the radial spring force. As a result of the frictional force reproduction described above, the change over force may achieve a higher value than is necessary for the actual synchronizing process. In this way, the desired shifting force reduction is achieved, but it is impossible to obtain an optimum power ratio.

b) Conical clutches are as a rule used as the synchronization coupling. In order for these to function without disturbance, it is necessary for both conical surfaces to run concentric with one another. As a result of the radial support of the locking effect by the synchronizing ring via the inclined locking surfaces of the pressure element, the synchronizing ring is prevented from centering with respect to the coupling cone which may lead to operational disturbances.

A further locking synchronization arrangement is known in which radially orientated pressure elements are also used. However these only have the function of transmitting the initial locking force to the synchronizing ring, whilst the main shifting force and in particular the locking effect is applied to the synchronizing ring by additional generally known devices. Apart from their different function, the pressure elements in this case also have the drawback that they perform a radial linear sliding movement during change over leading to a reproduction of frictional force (self-locking effect).

It is an object of the invention to provide an easier shift (i.e. a reduction of the shifting movement) by the use of an "internal shifting force ratio".

According to the invention, there is provided a locking synchronization arrangement for gear shifts, wherein two concentrically disposed components,

for example a gear shaft and gear wheel, rotating at different speeds may be firstly synchronized by means of a friction clutch and then positively connected in the direction of rotation, comprising a synchronizing body member which is rigidly connected to the shaft and has external clutch teeth on its outer periphery, a displacable annular shift sleeve disposed axially with respect to the body member and having internal clutch teeth engageable with external clutch teeth on the gearwheel which is disposed concentrically thereto, at least one synchronizing ring which is disposed in the axial direction between the gearwheel and the synchronizing body member and has a positive rotary synchronization with the synchronizing body and a friction surface which cooperates with a friction surface on the gear wheel, shift elements which are disposed between the synchronizing body and the shift sleeve, and which transmit the axially orientated shift force from the shift sleeve to the synchronizing ring and simultaneously lock the shift path of the shift sleeve until the synchronizing body and the gearwheel are synchronized, each shift element being a locking and pressure lever which is mounted in the synchronizing body so that it may not be displaced in a radial or Cardanic manner and may perform a free limited pivotting movement in all directions, the shift sleeve having locking surfaces or gaps therein.

Preferred embodiments of the invention will now be described, by way of example only, with reference to the accompanying drawings, in which:—

Fig. 1 is a cross-section (in the plane of rotation) through a shift element for an internal cone synchronizing ring;

Fig. 2 is a longitudinal section through Fig. 1;

Fig. 3 is a view of the locking gate of Fig. 1 from the interior of the shift sleeve;

Fig. 4 is a tangential section through the synchronous body with a radial external view of the pressure cage of Fig. 1;

Fig. 5 is a cross-section (in the plane of rotation) through a transfer element for an external cone synchronizing ring;

Fig. 6 is a longitudinal section through Fig. 5;

Fig. 7 is a view of the locking gate of Fig. 5;

Fig. 8 is a tangential section through the shift element of Fig. 5;

Fig. 9 is a longitudinal section through an automatic axial synchronizing ring adjustment.

Synchronizing body member 1 with a gear shaft positively connected thereto, shift sleeve 3 with internal clutch teeth 4, synchronous internal cone clutch positively connected with gear wheels Z1, Z2 and external clutch teeth 6 are generally known. In the case of an internal cone synchronization system, synchronizing ring 2 with its frictional surface – generally a cone clutch – is disposed inside the clutch teeth.

The invention relates to the synchronous shift element, comprising pressure lever 7, pressure cage 8, locking gate 15 and pre-locking ball bearing 10, which is designed to transmit the shift force applied from outside but only in the axial direction to the shift sleeve 3 in an amplified form to the synchronizing ring 2. At the same time, the frictional torque produced on the coupling cone and acting on the synchronizing ring 2 must be supported in the peripheral direction in such a way that a reverse coupling effect is created on the shift sleeve 3 as a result of which its axial path of displacement is locked as long as there is a frictional torque. The ratio: reverse coupling effect/axial shifting force must be stable during the entire synchronization process. The synchronous shift element may to a certain extent be seen as a "servo-pilot".

In order to maintain a constant equilibrium condition of all the force effects, at least two synchronous shift elements at the same angular spacing are disposed in the plane of rotation for a synchronization unit. A rectangular recess is provided in the synchronizing member 1 for each synchronous shift element.

The pressure lever 7 is generally cylindrical and has a tapered foot formed with a ball pin by means of which it is mounted to pivot in all directions in the synchronizing member 1 but the degree of pivotting is limited. At the end of the pressure lever (pressure lever head) and in the shaft area, a flange with a ball-shaped flange surface is formed in each case. The pre-locking ball bearing 10 is disposed in a coaxial open bore in the pressure lever head, the ball bearing being urged outwardly by means of a spring.

The pressure lever 7 is designed to effect power transmission to the synchronizing ring 2. Consequently high contact stresses occur in the support of the synchronizing ring. In order to maintain these within acceptable limits, a pressure cage 8 is provided between the pressure lever 7 and the synchronizing ring 2. This cage is mounted on the shaft flange of the pressure lever 7 and may be displaced in the radial direction over a limited angular movement. The pressure cage has, in the axial direction, a pressure surface 11 acting against the synchronizing ring and a respective pressure surface 12 in both peripheral directions. The pressure cage is radially supported against the interior of the synchronizing ring 2 by means of symmetrically disposed flanges 13. The flange surfaces 13 have the same curvature as the interior of the ring 2 and the pressure cage is urged radially against the synchronizing ring by means of springs 14. As at least two pressure cages 8 are provided with a uniform angular spacing from each other, the synchronizing ring therefore receives a resilient radial fastening.

The pressure lever head is located in the shift sleeve 3 by a locking gate 15. This comprises an axially orientated shift groove 16 having arranged centrally and symmetrically thereto, two locking gaps 17 in the peripheral direction. The base surface of the shift groove 16 and the locking gaps 17 is curved about an axis which is parallel to the gear axis and passes through the point of articulation of the pressure lever 7. A pre-locking groove 18 is sunk in this base surface and is arranged transverse to the shift groove 16 but inside the locking gaps 17. The flanks of the locking gaps 17 lie below a calculable angle to the peripheral direction. The locking gate 15 may be directly machined into the shift sleeve 3 by milling or by electrochemical treatment. As a separate compo-

nent it must be positively connected to the shift sleeve 3 in the axial direction and must be supported against tangential displacement either on the shift sleeve or on the synchronizing ring 2 (as shown in Fig. 1).

The operation of the illustrated embodiment is as follows:–

When the shift sleeve 3 is in its central position (see Fig. 2), the pre-locking ball bearing 10 is engaged in the pre-locking groove 18 of the locking gate 15. Both synchronizing rings 2 are axially spaced from the internal clutch cone 5 and rotate synchronously with the synchronizing member 1.

If the shift sleeve 3 is moved to the left in Fig. 2, the synchronizing ring 2 is urged against its internal clutch cone 5 via the pressure lever 7 and the pressure cage 8. As in general all the parts operate in oil, there is in the first instance a film of oil between both clutch components which should only be squeezed out if the friction coefficient is so small without solid body contact that no useful clutch friction torque could occur. As a result of the defined spring force on the pre-locking ball bearing 10, it is possible to apply an axial force, amplified by the lever action of the pressure lever 7, which is large enough on the synchronizing ring that a solid body contact takes place within a very short time (approx. 0.01 sec.).

As soon as a frictional torque builds up, the synchronizing ring 2 is rotated in the peripheral direction, is supported via the pressure surface 12 of the pressure cage 8 in the peripheral direction on the pressure lever 7 and pivots the latter into the locking gap 17 of the locking gate 15. The shift direction in each case is a result of the direction of the speed difference between both coupling components. The shift path for the shift sleeve 3 is therefore locked (Fig. 3). The axial force exerted on the shift sleeve then acts on the pressure lever 7 via the flank of the locking gap 17. For the stationary synchronization process with stable friction coefficients, there must be a force equilibrium at the locking gap 17. The axial transfer force F_a and a peripheral force F_u produced by the synchronizing ring then operates in a reduced manner with the transmission ratio of the pressure lever 7. The flanks of the locking gap 17 must be orientated at right angles to the resultant force direction F_s. The proportionality F_u/F_a is maintained until the synchronous coupling is synchronized. F_u is then smaller whilst F_a continues to operate. As a result of this a free force component arises at the inclined locking gap flanks in the peripheral direction, the pressure lever 7 then being pushed back into the centre of the shift groove 16 and the synchronizing ring rotated back through a small angle. The locking effect is therefore cancelled and the shift sleeve 3 may be axially moved on and engaged in the counter-clutch teeth.

In contrast to prior art pressure elements, the pressure element 7 is not radially displaceable and no radially directed forces of reaction operate on the pressure cage 8.

In the case of the stationary synchronization process, the force ratio F_u/F_a also remains constant at the pressure surfaces 11 and 12 of the pressure cage 8. It is therefore possible to provide an inclined pressure surface lying approximately at right angles to the resultant force direction instead of the right-angled arrangement of two pressure surfaces.

Figs. 5 to 8 show an extension of the invention in which the shift element is described in connection with external cone synchronizing rings.

In this embodiment the clutch friction surface lies outside the clutch teeth and the diameter of the friction clutch may be the same as or greater than that of the shift sleeve.

On account of the internally disposed clutch teeth, the radial height of the shift sleeve 3 is greater than in the embodiment of Figs. 1 to 4. For each shift element there is in this case provided a circular, inwardly open recess in the shift sleeve 3 and the synchronizing member 1 has only one low flat area.

The locking gate 15 is fitted in the external form of the recess in the shift sleeve. Axial fastening to the shift sleeve takes place by means of outwardly projecting stop flanges 15*a*. For tangential fastening, circular segments 21 are provided on both plane sides of the shift sleeve 3. The locking gate 15 must be long enough in the axial direction to ensure that in the case of use of the maximum shift path, the pre-locking ball 10 is still radially held.

The operation of the arrangement just described is as follows:–

On account of the large diameter of the synchronizing ring 2, it has, bearing against the pressure cage 8, oblique inwardly projecting cams 22 with pressure surfaces 23, 24 for transmission of the axial and tangential forces.

To further improve the operation, common centering of the shift sleeve in the shift element may be provided by means of a pressure cage spring 14 formed as a symmetrical arm leaf spring, e.g. the double U leaf spring illustrated. The lower spring arm lies on the synchronizing member 1 and is centered on the ball pin of the pressure lever 7 by means of a centering hole 14*a*. The upper spring arm has inclined centering surfaces 14*b* in the axial direction which bear against the pressure cage 8.

With the embodiment of Figs. 5 to 8, there is also the advantageous possibility of using disc clutches instead of cone clutches.

As a result of wear on the friction surfaces of the synchronous coupling, there is an increase in the axial shift path which is the more noticeable on the shift sleeve, the greater the pressure lever ratio is. In order to compensate for this effect, each shift element may be provided with an automatic axial synchronizing ring adjustment (see Fig. 9).

Peripheral grooves 25 are provided on the cylindrical internal surfaces of the synchronizing ring 2 on which the pressure cage flange 13 lies. These grooves have a saw-tooth profile and their steep flanks rest against the pressure cage 8 and have a very fine pitch (approximately 0.1 to 0.2 mm), but no thread. The pressure cage flange, which has the same peripheral curvature as the interior of the synchronizing ring, comprises at least one shaped edge 26 extending in the peripheral direction which engages in the groove 25 of the synchronizing ring. A U-shaped ring holder 27 whose radially positioned arms are adapted to the internal ring shape and also

comprise a profile edge are urged radially against both synchronizing rings by means of a spring element 28, preferably a leaf spring. The spring element 28 is in this respect independent of the pressure cage spring 14. The ring holder 27 is centered on the ball pin of the pressure lever by means of a centering hole, but with a predetermined axial clearance s2 which must be greater than the groove pitch. A clearance of more than one groove pitch must be provided in both axial directions between the ring holder 27 and the pressure cage 8. In order to provide improved centering of the ring holder 27 and the pressure cage spring 14, a sleeve 29 with a cylindrical external edge is provided for mounting of the pressure lever 7 in the synchronizing member 1.

In comparison with known locking synchronization systems with pressure element transmission, the invention provides a considerable advance in that instead of the linear sliding movement of the pressure element at the contact points with the highest pressure loading, there is an articulated rotary movement with substantially reduced frictional resistance.

In the known pressure element embodiments, the frictional resistance has a progressive nature with respect to the lever ratio and reaches a limit value (automatic locking). In contrast, the shift element of the invention has a diminishing type of frictional resistance with respect to the lever ratio.

This improvement provides the possibility of constructing the pressure lever transmission of any size which is not possible for physical reasons with the known constructions.

By means of the use of a pressure cage at the point of the highest pressure, loading optimum contact surfaces are provided for both the pressure lever and the synchronizing ring. This enables a free choice of material for the synchronizing ring.

No radial force effect is exerted on the synchronizing ring by the shift sleeve. This is radially guided in a resilient manner on the synchronous body via the pressure cage springs and may therefore be centered in an unhindered manner on the clutch cone. Moreover the synchronizing ring is also guided in all directions in the non-shifted condition by the radial initial spring tension and is unable to freely pivot or rock so there is no rattling of the ring.

With the substantially improved internal power ratio, it is possible to increase the cone angle (6-7°) of the synchronous coupling which may be used at present by several degrees. This step facilitates the automatic triggering of the clutch portions and additionally improves the shifting characteristics.

The embodiment with the outer cone synchronizing ring enables an optimum dimension of the clutch friction surface in a given installation area. If the effect of a large lever ratio of the pressure lever ($i \geqq 5$) is also used, it is therefore possible to increase the operating and torque loading of locking synchronization arrangements by a multiple which leads to the possibility of new applications.

This load increase (in the same installation area) provides in particular for gear shifts with gear units disposed upstream or downstream (lorries, tractors) a considerable improvement with respect to shiftability.

It would be possible to use this new locking synchronization arrangement for "heavy-shift" step gears (for example downstream gears).

As a result of the "internal power ratio" the outer shift components (articulations, shift rods, shift forks) are unloaded which leads to the reduced wear of these components and additionally provides an improvement of the external degree of shift efficiency.

CLAIMS

1. A locking synchronization arrangement for gear shifts, wherein two concentrically disposed components, for example a gear shaft and gearwheel, rotating at different speeds may be firstly synchronized by means of a friction clutch and then positively connected in the direction of rotation, comprising a synchronizing body member which is rigidly connected to the shaft and has external clutch teeth on its outer periphery, a displaceable annular shift sleeve disposed axially with respect to the body member and having internal clutch teeth engageable with external clutch teeth on the gearwheel which is disposed concentrically thereto, at least one synchronizing ring which is disposed in the axial direction between the gearwheel and the synchronizing body member and has a positive rotary synchronization with the synchronizing body and a friction surface which cooperates with a friction surface on the gearwheel, shift elements which are disposed between the synchronizing body and the shift sleeve, and which transmit the axially orientated shift force from the shift sleeve to the synchronizing ring and simultaneously lock the shift path of the shift sleeve until the synchronizing body and the gearwheel are synchronized, each shift element being a locking and pressure lever which is mounted in the synchronizing body so that it may not be displaced in a radial or Cardanic manner and may perform a free limited pivotting movement in all directions, the shift sleeve having locking surfaces or gaps therein.

2. A locking synchronization arrangement as claimed in claim 1, wherein each locking and pressure lever has a pre-locking ball bearing which cooperates with a pressure cage and a locking gate disposed in the shift sleeve, the locking gate being fixed in the axial direction and supported in the tangential direction either on the shift sleeve or on the synchronizing body, a shift groove in the axial direction through which the head of the locking and pressure lever may slide, the locking surfaces or gaps being provided in a central symmetrical arrangement, in which gaps the head of the locking and pressure lever may engage in the peripheral direction, the flanks of these locking gaps being set below by a predetermined locking angle φ with respect to the peripheral direction, a pre-locking groove with flanks inclined to the radial plane provided in the centre of the locking gate, with which groove the pre-locking ball bearing engages, the pressure cage having a central cylindrical bore by means of which it is mounted for limited angular movability on the locking and pressure lever and being displaceable coaxially with respect thereto, pressure surfaces in each peripheral direction on the

pressure cage, and flange surfaces disposed in the radial direction which cooperate with counter-surfaces on the synchronizing ring.

3. A locking synchronization arrangement as claimed in claim 2, wherein radially acting spring elements are provided between the synchronizing body and the pressure cage.

4. A locking synchronization arrangement as claimed in claim 2 or claim 3, wherein the pressure cage has an inclined pressure surface whose flange direction lies at right angles to the resultant force direction which results from the axial shift force and frictional peripheral force on the synchronizing ring.

5. A locking synchronization arrangement as claimed in any one of claims 2 to 4, wherein the synchronizing rings are disposed in the axial direction on both sides of the synchronizing body, the locking gate and the pressure cage being formed symmetrically with respect to the radial median plane and a reciprocal shift path being provided to each of the two synchronizing rings.

6. A locking synchronization arrangement as claimed in any one of claims 1 to 5, wherein the locking and pressure levers are uniformly peripherally distributed on the synchronizing body, the synchronizing rings having radial clearance with respect thereto.

7. A locking synchronization arrangement as claimed in any one of claims 1 to 6, wherein the locking gate is part of the shift sleeve.

8. A locking synchronization arrangement as claimed in any one of claims 1 to 7, wherein the synchronizing rings are formed with conical clutch friction surfaces on their exterior, the synchronizing rings having cams with operationally determined pressure surfaces relating to the pressure cage, said cams projecting radially inwardly and against the pressure cage of each shift element, a circular, inwardly open recess in the shift sleeve for each shift element, the locking gate having an external curvature which is identical to the recess in the shift sleeve and provided in the peripheral direction for axial support of the shift sleeve with contact ribs, support in the peripheral direction being provided by circular segments guided in circular grooves on both plane sides of the shift sleeve.

9. A locking synchronization arrangement as claimed in any one of claims 3 to 8, wherein each spring element which presses the pressure cage outwardly against the synchronizing ring is formed as a U-shaped arcuate leaf spring, the leaf spring being centered on the foot of the pressure lever for the common centering of the shift element and provided with inclined surfaces with respect to the pressure cage in the axial direction.

10. A locking synchronization arrangement as claimed in any one of claims 1 to 9, wherein the cylindrically curved interior of the synchronizing ring on which the pressure cage of each shift element lies has grooves in the peripheral direction of a very fine pitch and with a saw-tooth profile, the steep flanks of said grooves being orientated towards the pressure cage, a flange surface of the pressure cage having for each synchronizing ring at least one projecting profile edge corresponding to the groove profile on the synchronizing ring, a U-shaped ring holder whose arms are orientated in the radial plane having a profile edge which corresponds to the groove profile on the synchronizing ring, the ring holder being centered with a predetermined clearance s2 in the axial direction on the ball pin of the locking and pressure lever by means of a centering hole, a radially and outwardly acting leaf spring or diaphragm spring being provided between the synchronizing body and the ring holder, which spring is independent of the spring element of the pressure cage.

11. A locking synchronization arrangement as claimed in any one of claims 1 to 10, wherein a sleeve is provided for the Cardanic mounting of the locking and pressure lever in the synchronizing body and for the simultaneous centering of the pressure cage and the ring holder.

12. A locking synchronization arrangement substantially as herein described with reference to the accompanying drawings.

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